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A NOVEL DESIGN OF AN INJECTOR/IGNITOR UNIT FOR METHANOL FUEL IN A HEAVY-DUTY ENGINE

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RÉSUMÉ

En novembre 1988, Énergie, Mines et Ressources Canada a fait une demande de propositions quant à des projets qui seraient menés dans le cadre du Programme d'aide à la recherche relative aux carburants de transport (ARCT). Ce programme avait pour but d'encourager la recherche à haut risque dans le domaine des carburants de rechange. À l'époque, la société SYPHER:MUELLER International gérait le projet Méthanol pour les moteurs puissants (projet MPMP) pour le compte d'Énergie, Mines et Ressources Canada. Ce projet comportait la mise à l'essai en vraie grandeur du méthanol comme carburant d'autobus et de camions lourds. L'un des principaux problèmes mis en évidence par le projet MPMP avait trait à la très courte durée utile des injecteurs de carburant employés dans les moteurs alimentés au méthanol. Trois grands motoristes ont fourni des moteurs aux fins du projet : Caterpillar Incorporated, Cummins Engine Company et Detroit Diesel Corporation. Tous les moteurs étaient munis de systèmes classiques de type diesel pour l'injection mécanique par pression du carburant dans la chambre de combustion. Les seuls changements ont été apportés aux spécifications matérielles afin de tenir compte des propriétés chimiques différentes du méthanol, en plus d'augmenter la capacité d'écoulement du carburant pour compenser le contenu énergétique moins élevé. Des dépôts carbonés durs ont bouché les orifices de l'injecteur après une très courte période de temps, souvent après moins d'un mois ou 5 000 km de service. Normalement, un injecteur de diesel peut fonctionner plusieurs années ou au moins permettre de parcourir de 200 000 km à 300 000 km avant de s'encrasser de dépôts ou de nécessiter de l'entretien.

Des recherches effectuées au début du projet MPMP sur la durée utile des injecteurs ont montré que les injecteurs de méthanol utilisés dans une unité d'essai spéciale ne répondaient pas aux critères au moment de vérifications périodiques de la pression et du débit, et ce après des périodes d'essai beaucoup plus courtes que la durée de service dans les véhicules. Très peu des injecteurs ont eu une durée utile de plus de 50 heures dans l'unité d'essai où l'on simulait l'utilisation d'un moteur à une vitesse constante de 1 000 tr-min. Cette situation porte à croire que bon nombre des injecteurs qui fonctionnent encore dans les véhicules sont inutilisables selon des critères normaux applicables aux moteurs diesel, mais qu'ils sont capables de fournir un rendement et une économie de carburant acceptables sans produire de gaz d'échappement visibles. Le projet MPMP et d'autres essais en vraie grandeur du méthanol utilisé comme carburant ont montré par la suite que les injecteurs de méthanol peuvent se détériorer au point de produire des fumées abondantes dans les moteurs diesel, sans donner de signes externes de leur état. Par conséquent, on a formulé une hypothèse selon laquelle

les exigences de l'injection de méthanol pourraient être moins draconiennes que celles de l'injection de carburant diesel, et qu'il serait possible d'avoir recours à un injecteur de conception plus simple.

Étant donné la situation, la possibilité s'est présentée d'examiner l'injection du méthanol dans les moteurs lourds. La SYPHER:MUELLER savait qu'une société mandataire qui fabriquait un modèle amélioré d'un moteur d'automobile à deux temps, alimenté à l'essence, avait mis au point un nouveau type d'injecteur de carburant. Cet injecteur fait appel à l'air sous pression pour souffler un jet de carburant dans la chambre à combustion du moteur, par opposition à un système diesel ordinaire qui a recours une pompe à mouvement rectiligne. L'injecteur semblait être beaucoup plus simple à construire qu'un système diesel, et ses tolérances et son jeu interne étaient beaucoup plus élevés. On a pensé que ces détails le rendrait moins sujet à l'encrassement pas des dépôts de combustion, et plus facile et moins dispendieux à fabriquer qu'un système diesel ordinaire. La SYPHER:MUELLER a donc proposé un projet visant à examiner la faisabilité d'utiliser ce genre d'injecteur dans un moteur lourd alimenté au méthanol.

Le projet avait pour but d'obtenir un des injecteurs brevetés, de l'installer dans un moteur d'essai semblable à un des moteurs utilisés aux fins du projet MPMP, et d'entreprendre un programme de vérification du rendement afin d'évaluer la possibilité d'utiliser le principe de l'injection par jet d'air dans un modèle pratique de moteur lourd alimenté au méthanol.

La SYPHER:MUELLER International a retenu les services de la société ORTECH International pour fournir le moteur et les installations d'essai et effectuer des essais de rendement. La ORTECH avait participé à des travaux très semblables entrepris par l'Université de Toronto, relativement à l'injection de carburants gazeux comme l'hydrogène et le gaz naturel dans un moteur à allumage par compression.

Il était prévu que si la société qui avait mis au point l'injecteur par jet d'air était dans l'incapacité de fournir une unité d'essai pour le projet, ou refusait de le faire, la ORTECH construirait un injecteur semblable en s'inspirant des connaissances acquises au cours des travaux menés par l'Université de Toronto et des données publiées sur l'injecteur breveté. Au début du projet, il a été convenu qu'il serait beaucoup plus souhaitable d'obtenir une unité brevetée, dans la mesure du possible, afin d'éviter des complications et des dépenses imprévisibles. Cependant, les négociations avec le fournisseur se sont prolongées alors qu'on ne s'y attendait pas et le projet a été retardé bien au-delà du délai prévu; il a donc fallu modifier la durée et la portée du contrat.

I INTRODUCTION

A. BACKGROUND

In November 1988 Energy, Mines and Resources issued a request for proposals for projects to be carried out as part of the Transportation Energy Research Assistance (TERA) program. The intention of the program was to encourage high-risk research work in the area of alternative fuels for transportation purposes. At the time SYPHER:MUELLER International was managing the Methanol In Large Engines (MILE) project for Energy, Mines and Resources. This project was a large scale field trial of methanol as a fuel for heavy-duty trucks and buses. One of the primary concerns which had emerged from MILE was the very short service life of the engine fuel injectors when burning methanol fuel. Three major engine manufacturers had supplied engines to the project, Caterpillar Incorporated, Cummins Engine Company, and Detroit Diesel Corporation. All of the engines retained the standard diesel-type high pressure solid injection systems to inject the fuel charge into the combustion chamber. The only changes made involved material specifications to cope with the different chemical properties of methanol, and an increase in fuel flow capacity to allow for its lower volumetric energy content. The injector orifices became blocked with hard carbonaceous deposits after a very short period, frequently less than 1 month or 5,000km of fleet service. A diesel injector normally lasts several years or at least 200,000 km to 300,000 km before any significant build up of deposits occurs, or any form of maintenance is necessary.

Some research work on injector life undertaken during the early part of project MILE showed that methanol injectors operated on a special test rig failed routine pressure and flow rate checks after test durations far shorter than actual in-vehicle service times. Few injectors lasted more than 50 hours on the test rig, which simulated engine operation at a steady 1,000 rpm. This led to the speculation that many of the injectors still working in vehicles were in fact unserviceable by normal diesel criteria, but still capable of providing acceptable performance and fuel economy together with no visible exhaust emissions. Evidence from MILE and other methanol field trials has subsequently shown that methanol injectors can deteriorate to levels which would produce drastic smoking in a diesel without any corresponding external evidence of their condition. This is turn produced an idea that perhaps the requirements of methanol injection were less exacting than those of diesel, and could be met by a simpler type of injector unit.

In view of this situation there was scope for investigation into the question of fuel injection in methanol heavy-duty engines. SYPHER:MUELLER was aware of a new type of fuel injector which had been developed by a proprietary company building an improved form of two-stroke gasoline automobile engine. This injector employed high-pressure air to blast a fuel spray into the engine combustion chamber, in contrast to the positive displacement pump action of a normal diesel system. It appeared to be considerably less complicated in construction than a diesel system, with wider internal tolerances and clearances. It was speculated that these features might make it less prone to clogging by small amounts of combustion deposits. It was also expected to be much easier and cheaper to manufacture than a normal diesel system. SYPHER:MUELLER therefore proposed a project to investigate the feasibility of using this type of injector in a heavy-duty methanol engine.

The intention of the project was to obtain one of the proprietary injectors, install it in a test engine similar to one of the engines in use in project MILE, and to carry out a program of performance testing to assess whether the air-blast injection principle could be used in a practical heavy duty methanol engine design.

SYPHER:MUELLER International engaged ORTECH International of Mississauga to provide the test engine and test facilities, carry out the adaptation and installation of the injector, and run the performance tests. ORTECH staff had been involved in closely related work done by the University of Toronto, concerning the injection of gaseous fuels such as hydrogen and natural gas into a compression ignition engine.

In the event that the company which had developed the air-blast injector was unable or unwilling to supply a test unit to the project, ORTECH was to construct an injector along similar lines using its knowledge gained from the U of T work, and published data relating to the proprietary injector. It was agreed early in the project that it was much more desirable to obtain a proprietary unit if at all possible, in order to avoid unpredictable project complications and expense. However, the negotiations with the supplier became unexpectedly lengthy, with the result that the project was delayed far beyond the original time-frame, and a number of amendments to the contract duration and scope became necessary.

B. OBJECTIVES

The original overall objective of the project was as follows:

To discover whether the air blast injector concept could be used successfully in a conventional design of heavy-duty engine operating on methanol fuel.

Five sub-objectives were identified:

1. To procure a proprietary injector and modify it to a size and configuration suitable for operation in a test two-stroke Detroit Diesel methanol engine.

2. To run a series of baseline comparison tests on the test engine using standard diesel fuel to establish normal performance levels.

3. To fit the modified injector into the test engine and to provide necessary hardware and engine modifications to enable operation on methanol fuel.

4. To repeat the test sequence as used in sub-objective 2 and record the results achieved

5. To analyze and compare the performance figures obtained, particularly in respect of power and torque development, exhaust emissions levels, and deterioration of injector performance over time.

As the project progressed it became apparent that an unforeseen amount of time and effort was being absorbed by negotiations for the supply of a test injector, and later on by the fitting of the injector to the test engine. Although the overall objective of the project remained unchanged the work planned in paragraph 2 above did not take place. In order to save time and expenditure existing data on the Detroit Diesel engine was used for comparison. In addition, the testing program was severely restricted in scope by the eventual difficulties experienced in trying to achieve reliable operation of the engine/injector combination. This meant that considerably less data than planned was obtained. It also effectively curtailed any plans to encourage other interested parties such as the engine manufacturers to contribute to the project, and the hope of immediately generating any commercial "spin-off" effects.

C. METHODOLOGY

The conduct of the project was divided initially between SYPHER:MUELLER International and ORTECH International. At a later stage it became necessary to develop a close working partnership with the injector manufacturer, which provided a considerable amount of assistance in the adaptation and fitting of the injector to the engine. The manufacturer requested that it should not be mentioned by name in any report as a condition of its participation in the project. The name of the company was disclosed to both the Scientific Authority and the SSC Project Officer during the course of the project.

SYPHER:MUELLER was responsible for overall project management, negotiations with the injector supplier, investigation of parallel work, liaison with outside parties and assessment of commercial possibilities. ORTECH undertook all engine and test bed configuration, injector installation and test operation, and interpretation of results.

The test engine used was a two cylinder Detroit Diesel 2-71 two-stroke unit, similar in design to the Detroit Diesel engines used in the MILE project buses at Medicine Hat and Winnipeg. These particular engines had proved most prone to injector plugging problems during MILE operations. In addition, the air blast injection system was better adapted to the two-stroke engine configuration. A normal diesel injection system produces in excess of 10,000 psi pressure in order to force a spray of fuel into the engine combustion chamber against peak compression pressure. The air blast system used was only capable of pressures in the 100 psi range. It was therefore necessary to inject the fuel charge early in the compression stroke before any appreciable back pressure was built up. Both the operational cycle and the construction of the Detroit Diesel engine lent themselves to this modification better than the more common four-stroke engine configuration. Injection of the fuel so early in the stroke would be impossible with diesel fuel, because spontaneous ignition would occur long before top dead centre. However, it was feasible with methanol because of the far higher temperature needed for ignition. The test engine used a spark plug as an ignition assistance source, and it was expected that ignition would not occur until the spark plug was energized. In fact, pre-ignition of the methanol proved to be the most severe limitation on engine operation when testing got under way.

D. CHARACTERISTICS OF THE INJECTOR

The proprietary injector was designed to inject gasoline directly into the combustion chamber of a lightweight two-stroke automobile engine shortly before top dead centre, for subsequent ignition by a spark plug. It replaced a conventional carburetor or port fuel injection system, and avoided the direct loss of unburned fuel from the transfer to the exhaust port. This has always been a particular problem with a crankcase scavenging two-stroke, resulting in unacceptably high exhaust emissions by modern standards, and poor fuel economy. Because it was only necessary to pass pure air through the crankcase instead of a fuel/air mixture the use of this injector also allowed an improved lubrication system. In an ordinary two-stroke gasoline engine it is impossible to achieve complete segregation of the engine lubrication oil and the gasoline fuel, as is the case in a four-stroke engine. Indeed, this type of engine is frequently lubricated by mixing oil in with the gasoline supply, which has a disastrous effect on exhaust emissions both visibly and invisibly. The two-stroke gasoline engine has therefore almost completely disappeared from the automotive field except for small numbers of motorcycles, although it remains widespread in chain saws, lawn mowers, snowmobiles and other types of small vehicles and equipment.

The introduction of the direct injection fuelling principle produced a drastic transformation of engine emissions characteristics. This in turn opened the door to exploitation of the advantages of the two-stroke engine, which include high power to weight ratio, simple construction, and consequent low cost.

It is pertinent to mention at this stage that there are a number of configurations in use for engines operating on the two-stroke cycle (one power stroke every revolution), and that a two-stroke diesel engine is unlike a two-stroke gasoline engine. The diesel does not pass the inlet air stream through the engine crankcase and uses high-pressure direct fuel injection of necessity in order to operate. It therefore does not incur the same problems with unburned fuel loss or combustion of engine oil as the gasoline engine, although careful design is necessary to maintain its equality with a four-stroke diesel engine in these respects.

II EXPERIMENTAL WORK

A. TEST ENGINE

The test engine selected was a two-stroke engine, Model 2-71 manufactured by Detroit Diesel Corporation (DDC). It used a mechanically driven blower to force induction air into the cylinders, but was not fitted with a turbocharger. The displacement per cylinder was the same as the 6V-71, a popular bus engine in Canadian Transit applications. The two-cylinder configuration allowed either operation of one or two cylinders on the new fuel system. Being a production engine, engine parts, if required, would be readily available. The detailed specifications and ratings of the diesel production version are given in Appendix A. The stock compression ratio of the engine was 17:1. This was lowered to 13:1 by reducing the length of the connecting rods. Although the nominal compression ratio of the engine was 13:1, the actual compression was somewhat lower, in the region of 11 or 12:1. This actual CR varied with speed of the engine due to inlet port configuration and air inertia effects. The cylinder head of the engine had two exhaust valves per cylinder. This was a distinct advantage, since it allowed more room for the injection and spark plug location in the cylinder. The test engine was set-up at ORTECH International in engine Test Cell #5, and connected to an eddy current dynamometer.

B. FUEL SYSTEM ADAPTATION

Two approaches were evaluated for injecting the methanol into the engine. The first was to inject the methanol into the cylinder though the cylinder liner intake air ports. The second was to inject through the cylinder head essentially replacing the diesel injector. The first approach put a strict limitation on the window of injection, because the piston covered the intake ports and prevented further injection long before cylinder pressure would stop injection. Due to this limitation, the second option was pursued, and the injection of the methanol was arranged to take place through the cylinder head of the engine. The air-blast injector required manifolds to feed it air and fuel. These manifolds were produced by designing and fabricating concentric adapter sleeves which had passages drilled to supply fuel, air and electrical connections. These adapter sleeves are shown in Exhibits 2 and 3.

The adapter sleeve fitted into the head where the diesel injectors would normally be placed. Due to their slightly larger diameter the copper sleeves in the head had to be removed and the passage widened. This made the outer adapter sleeve the seal for the water jacket in the head. This had the advantage of delivering better cooling to the injectors and fuel manifolds, but made the fit at the top surface and the fire deck of the head critical. The entire assembly, including the injectors, was held in the head by two hold down clamps bolted to the head. The inner adapter sleeve was designed to protrude slightly above the outer sleeve, head surface and the injector top surface. This allowed the hold down clamps to exert a force on the outer sleeve, transmitted through the inner sleeve and the lower injector, at the point where the outer sleeve sealed with the lower part of the water jacket.

The spark plugs were placed though the head and water jacket, and entered the combustion chamber approximately two thirds of the piston radius out from center. Due to space limitations, it was not possible to make a separate spark plug sleeve to seal the water jacket. Instead, the spark plug and sleeve had to be made as one unit. This required machining the flats of a spark plug, making a close fitting hex headed tube and welding this tube to the spark plug. The taper seal of the spark plug was the seal at the lower end, while an O-ring seal secured the water jacket at the top.

A cross section of the cylinder head is given in Exhibit 1, showing the injector and spark plug position, as well as the position of the combustion chamber pressure transducer. Exhibit 4 shows a top view of the cylinder head with one injector installed. The supply lines for air and fuel, together with the wiring to the injectors and spark plugs, are shown in Exhibit 5.

The air flow to the engine was initially controlled with a blower bypass. This allowed air which had been drawn in by the blower to be directed from the airbox back to the intake pipe rather than into the cylinders. The disadvantage of this form of air control is that the recirculated air is elevated in temperature by the work done on it by the blower. Later in the program, a throttle plate was used as a means of air control. This gave a wider range of air flows but introduced throttling losses to the engine operation.

Information from patent documents indicated the air blast injector was a single unit device. Initial work on a controller for injection and spark timing based on this information was started. After contacting the manufacturer and coming to an agreement on terms of use, it was found that the device was actually a two component per cylinder system. This meant that the previous work done on the controller was unsuitable for the actual device to be used. Further negotiations with the manufacturer resulted in it allowing the lease of an experimental controller for the duration of the project.

The Engine Control Module (ECM) for the fuel and ignition system received two signals from the crank angle encoder mounted on the engine. One signal was a top dead center (TDC) reference marker, the other was crank angle degree. The ECM required one pulse per degree, and a TDC marker inverted from the encoder output. A conditioning board was built which translated the encoder signals into the format the controller required. The outputs of the ECM were TTL signals for injector actuation and spark triggering.

The injector signals were sent to an Injector Drive Module (IDM) which supplied the required current wave-form to operate the injector. The ignition signals went to the Bosch ignition modules which fired the appropriate ignition coil and produced the required dwell angle control for the ignition system. The ECM allowed specification of the injector fill and firing and duration, as well as ignition timing.

After fabrication of a wiring harness and installation of the injectors, the ECM signals were checked on a scope. The injector and ignition timing signals were unstable and varied in timing by plus or minus 8 to 10^{o} crank angle. The duration of the signals was correct. It was found that the description given for the encoder input signals was incorrect, and after adjusting the harness for this correction the injection signals became stable. The ignition signals were still in error. After consulting with the manufacturer schematics of the circuit were sent from which to troubleshoot the circuit. When the signal produced from the rotary encoder was compared to the signals generated by the ECM's internal circuitry it was found that the encoder signals needed further conditioning. A buffered inverter circuit was made with a one shot pulse width narrowing component to properly condition the input signals from the engine's rotary encoder. This corrected the problem. However, after a short period of operation the injectors ceased operation. It was found that the encoder signal was not being received by the ECM due to a failure of the input receiver chip in the ECM. The chip was replaced. It was then discovered that incorrect output

levels were appearing on the TTL line to the injector driver module, thus preventing injection. The output line drivers were replaced and the signals were checked and found to be within specification. The injector driver module output transistors were also checked and it was found that one output transistor off-state voltage was low at 9 VDC. After replacement, each component was checked individually. The ECM outputs, the IDM outputs and the injector operation sub-components checked out correctly. However, when the system was reconnected and operated the engine failed to fire. The injector line voltages were measured and were found to be improper. It was also found that the above mentioned input receivers and output drivers had failed again. It was decided that the problem resided in the IDM. No documentation was available to properly troubleshoot and repair it, so it was replaced with simpler foldback current driver transistors. This worked well and there was no experience of any further problems with the injector or ignition driver circuits. A schematic of the electrical set up is shown in Exhibit 6.

The fuel system was a pressurized closed loop recirculating type. Fuel was drawn from a reservoir on a digital mass scale by the in-line fuel pump. The fuel pump pressurized the fuel to approximately 90 psig and circulated it through the injector adapter sleeves, a pressure regulating relief valve, and then back to the fuel reservoir. The dry side of the pressure regulating relief valve diaphragm was supplied with a static biasing pressure to control the fuel system pressure. When the fuel system was first put in place the single fuel pump was incapable of delivering any flow at the required pressure. A second pump was placed in series with the first to reduce the pressure loading on the pumps. This system performed well at onset of testing, but over time, performance began to deteriorate. One pump failed after a few hours of testing. This was thought to be due to the system's inability to balance the load between the two pumps in series. After replacement of the failed pump the pressure to the injectors became erratic, varying between 70 and 100 psig. It was believed that the methanol had affected the diaphragm or valve seat of the pressure regulating relief valve causing sporadic operation. The pressure regulating relief valve was replaced, but the resulting fuel pressure was found to be too high or too low when the air blast was adjusted. Assuming that the flow produced by the replacement pump was too high for the regulator, two regulators were placed in parallel. This appeared to solve the problem, but erratic pressures returned later in the testing. It was found that pressure fluctuations occurred during cranking of the engine, thus voltage variations of the engine's electrical system may have been one of the contributing factors to the fuel pressure problem.

The engine was started and ran roughly on one cylinder. A leak from cylinder #1 into the water jacket was discovered. The lower seal at the #1 spark plug and the #1 outer injector were identified as the culprits. The outer injector sleeves were machined to accept O-rings, and the spark plug seat was remachined. The engine was reassembled and restarted. Operation was still erratic and firing shifted from #1 to #2 cylinder at times. On tear-down, the spark plug boot showed signs of arcing from the cable to the plug insert shell. This would explain the erratic performance of the engine. Better fitting spark plug boots were installed and appeared to solve the problem. Concurrently, it was found that the #1 fuel injector had developed a pinhole leak at its upper surface. Attempts to repair the leak were unsuccessful. A new injector had to be ordered.

C. TEST MATRIX

The test matrix was designed to investigate key variables which determine engine operation. Over the speed/load range covered, the spark ignition timing was to be varied. The most critical aspect to be investigated was the fuel introduction, comprising fill start, fill duration, injection start and injection duration, the latter two taking place during the second half of the scavenging process and early in to the compression stroke.

Exhibit 1 identifies the test matrix, the speed and load settings, and the variables to be investigated at each setting. A total of 77 test points was identified.

D. TEST RESULTS

As already mentioned, numerous problems were encountered with the engine set-up, the injectors and driver electronics. Due to the heavy reliance on an outside supplier, delivery times were difficult to control.

The engine was brought to an operational level and a total of 9 test points were run, with a complete set of data being collected for each. The data are summarized in Exhibit 2.

The engine was operated at a number of times between component failures. The operation was erratic and uncontrollable. On cranking, the engine would not fire until the ignition timing was advanced to 45 to 50 crank angle degrees before top dead center, (BTDC), and the fuelling level very high.

Once the engine caught, the firing would be rough and uneven, with just enough power to maintain running at approximately 300 to 500 RPM and no load. Lowering the fuelling rate would sometimes have an effect on operation, but at other times no noticeable effect would be seen, even when the fuel level was one third the starting level. Quite often at this point the engine would be operating on only one cylinder. After a minute or so of this operation the engine would suddenly surge in speed and smooth out in firing. The RPM would then rise to that set on the dynamometer and load would start to develop. At times this smooth operation could be maintained and some control over speed and load could be exercised, but often, for no apparent reason, the power would fall off and the engine would either die completely or resume rough and uneven idle.

At times, when operation was stable, the timing as set on the ECM had no effect on start of combustion. Retarding the spark to after top dead centre, or even pulling the spark plug high tension lead off the coil, would not stop the engine from firing. It became apparent that a serious pre-ignition problem existed.

In an attempt to prevent this pre-ignition a number of modifications were made to the engine. A fuel cooler was added to reduce fuel temperature and an external marine air charge cooler was put on the air inlet, with city water as a coolant. This could only be used as a precooler of the air charge because of its size, and could not be used to cool the hot recirculated air. The thermostat was removed and the heat exchanger setting lowered to reduce coolant temperatures. An external oil cooler was added to reduce the oil temperature and cooler running plugs were installed. None of these modifications had any effect on the pre-ignition problem. After a few hours of testing the coolant boiled over, indicating a leak of combustion gas into the coolant. Removal of the new spark plugs showed a leak had developed at the point where the extender tube was welded to the plug. Repair attempts resulted in the breakdown of the ceramic, and the old plugs were reinstalled.

Engine operation could not be maintained in a sufficiently controlled manner to attempt any mapping or emission measurements of the engine. During periods of operation the amount of control over the combustion was minimal at best. Even with significant changes in fuel injection timing and duration control of the engine always seemed more a matter of chance than design.

Speeds covered were from 850 to 1800 RPM, and loads from 4 to 122 ft.lbs torque. Power and efficiency ranged from 1 HP at idle, to 35 HP at 1800 RPM, full load, and up to 25 percent at 1350/1425 RPM, or peak torque, respectively. The cylinder pressure recordings are shown in Exhibits 7 through 9. The traces look quite smooth and are indicative of a homogeneous charge spark ignited engine. Further analysis of the heat release rates and combustion duration might be instructive.

III DISCUSSION OF RESULTS

The project did not wholly meet its objective. The numerous technical difficulties encountered, although individually small, effectively prevented sufficient engine operation to draw good conclusions about the feasibility of the air blast principle. In particular, it was impossible to run the engine for a period of time remotely approaching that required to make any useful statement about injector longevity. Specific problems included failure of the electronic control module, a coolant leak around the injector to cylinder head adaptor, and a compression leak through the injector. All of these were cured during the engine test period, but the time lost could not be regained. The most serious and significant problem was pre-ignition of the methanol fuel. Several attempts to improve the engine cooling failed to eliminate this phenomenon, which made controlled long duration engine operation impossible.

In a conventional diesel engine the start of ignition is controlled by the fuel injection timing. The methanol engines running in MILE vehicles controlled ignition initiation by the same method, although the Caterpillar and Detroit Diesel Engines both provided extra cylinder heating assistance to ensure that the fuel would ignite when injection took place. The third type of project engine, Cummins, employed an ignition enhancement additive in the methanol to provide similar auto-ignition properties to diesel fuel. In contrast the project engine operation was analogous to a gasoline engine, in that the fuel was introduced into the cylinder before the requisite point of ignition, and eventually ignited by a spark plug. If at any time during the compression stroke the mixture temperature rose above the auto-ignition point of methanol then pre-ignition would occur. There were several possible ways for this to happen.

General high engine temperature.

Localized high engine temperature or in-cylinder "hot spot"

High inlet air temperature.

General high engine temperature, if not caused by a specific fault, is associated with the capacity of the engine cooling system. During the project various measures were tried to increase heat dissipation from the engine without success. In an actual practical engine design the cooling system capacity may obviously be adjusted to suit the application. This was not possible within the time, budget and engineering constraints of the project.

Localized engine temperature problems and the possibility of "hot spots" within the cylinder could not effectively be addressed within the scope of the project. These phenomena are caused by the internal construction and geometry of the engine, and must be cured through careful detail design changes. The installation of the injector and spark plug may have contributed to the formation of hot areas within the cylinder. Again, in an actual practical engine this matter could easily be addressed at the design stage. "Hot spots" are not desirable in any engine and can readily be eliminated.

The inlet air temperature may easily be adjusted by use of a charge air cooler. (Aftercooler or Intercooler) The Detroit Diesel methanol vehicle engines are fitted with aftercoolers in addition to the unique exhaust gas control and blower air by-pass arrangements. These features provide a far greater level of control over inlet air conditions than could be achieved with the laboratory test engine conversion. If time and budget had permitted the next step in the project would have been the refinement of the aftercooler fitted to the test engine to handle recirculated blower by-pass air. All of these potential cooling problems could be eliminated by further engine development work.

The project test work did succeed in demonstrating that it was possible to inject methanol fuel into the engine using air pressure alone. At the start of testing it was not known whether it would be possible to introduce sufficient fuel to generate a useful power output within the time constraints imposed by the build-up of back pressure in the engine cylinder. The metering limitations of the injector, and hence its ability to control power output in the manner of an orthodox injection system, could not be adequately explored because of the erratic ignition performance. However, the pre-ignition difficulties experienced during testing were probably not due to the action of the injector, although the fuel mixing pattern would have had an influence on the in-cylinder conditions after the fuel was injected. If it is assumed that the pre-ignition could be eliminated by some of the measures considered earlier, then it would appear that the air-blast injector is a practical way in which to get fuel into this type of engine. Furthermore, it would not be difficult to adapt the design for much higher air pressures, say in the 500psi range. This would enhance the precision and accuracy of the spray process, in both the metering of fuel quantity and the positioning of the fuel spray in the cylinder, with beneficial effects on the combustion process. This overall premise

suggests that further test work would be profitable. The alternative is to accept that the pre-ignition phenomenon is uncontrollable, and that the concept of injecting the fuel ahead of the required ignition time is fundamentally unsound. In this case the air blast technique is ruled out, because a system capable of developing pressures to duplicate the action of an orthodox mechanical system would be impractical. A third point of view is to consider the construction of a low-pressure mechanical system instead of the air blast for injection at the start of compression.

IV COMMERCIAL CONSIDERATIONS

When the project was planned it was hoped that successful trials would evoke some interest in the air blast principle from diesel manufacturers. It was also hoped that the manufacturer of the injector, which has licensing agreements with a number of prominent automobile companies, might take the opportunity to promote the use of its product in a new application. The lack of success from the engine trials undermined these plans, and the unexpectedly long duration of the project also caused a certain loss of topicality in the subject which it was designed to address.

The project was conceived in late 1988, at which time the problem of fuel injector life in project MILE was very acute. Project vehicles had been through protracted periods of downtime, sometimes months, because of failure of existing injectors and lack of spares. Since that time Detroit Diesel Corporation has expanded its methanol engine research program enormously, and one of the priority subjects for examination has been injector life.

Detroit Diesel continued with the electronically controlled version of its high pressure unit injector, based on the design used ever since the first Detroit Diesel engines were produced in the 1930's. It attacked the tip plugging problem through measures including special engine lubricating oils, tighter control of oil passing the piston rings and entering the injector operating plunger, tip orifice size and number, and additives to the methanol fuel. The two most effective antidotes proved to be special low-ash engine oils, which were already in use when this injector project began, and a fuel additive developed by Lubrizol Corporation which was allowing almost tenfold increases in average injector life by the time this project concluded. Detroit Diesel felt that it was well on the way to discovering the solution to the problem without any change to its injection system.

The reason for the continuing project emphasis on Detroit Diesel engines relates to the acuteness of the injector problems experienced with these engines in project MILE, and the suitability of the design for testing the low-pressure air-blast injector. The Detroit Diesel is the only quantity produced two-stroke diesel truck and bus engine in North America, and the only two-stroke diesel manufacturer in the world planning to market methanol engines. The two-stroke diesel vehicle engine is equally uncommon in the rest of the world. Most manufacturers produce four-stroke engines, and the

pressure cycle of the four stroke engine does not allow low pressure in-cylinder injection with the same facility. It would be necessary to employ much higher air blast pressure to use the injector in a four stroke engine, entailing major re-design work and an entirely separate research and testing program. Any achievements with the two-stroke engine would be of little interest or value to a four-stroke engine designer.

Apart from the contrast between two-stroke and four-stroke engines there is the question of involvement in methanol engine development. Of the three manufacturers involved in MILE, Caterpillar, Cummins, and Detroit Diesel, Detroit Diesel is the only one to have made any degree of commitment to methanol. Cummins has diverted its alternative fuels work to natural gas. Caterpillar is operating one vehicle in a new fleet trial in California, but is not engaged in any large scale fundamental research work. In any case injector life in Caterpillar methanol engines has not been so unsatisfactory in all trials undertaken to date. Approximately 1/10 of the normal expectation from a diesel nozzle has been attained, and the company expects that comparable life to a diesel could be attained with reasonable amount of development work if it ever chooses to initiate production of a methanol engine.

Cummins methanol engines, using "Avocet" ignition enhancer in the fuel, have also been largely free of injector tip plugging, although the amount of fleet service by the Cummins engines is not as great as that for the Caterpillar and Detroit Diesel Engines.

Other prominent manufacturers testing methanol engines in North America include Navistar International, and Ford Motor Company. Both of these companies manufacture four-stroke engines. The German M.A.N. company was a pioneer in heavy-duty methanol engines for the North American market, but did not expand on the scope of its two initial fleet demonstrations in Los Angeles and Seattle, and is now no longer promoting methanol engines in the U.S.A. or Canada.

The injector manufacturer was extremely helpful once it had agreed to become involved in the project. However, it sees no commercially exploitable value in the work performed so far. It is also fully involved at present in the commercialization of the injector for the passenger automobile market, and has no desire or need to divert its engineering and design effort away from that objective at the present time.

A principle advantage of the injector in commercial terms would be its low cost in comparison to conventional diesel type injection equipment. The typical diesel system incorporates a positive displacement pump to generate the requisite injection pressure, a nozzle and restrictor valve to admit the fuel into the engine combustion chamber, and a fuel metering system to control the amount of fuel admitted and hence the engine power output. Because of the very high pressures (over 10,000psi), and the very small quantities of fuel involved, the equipment has to be manufactured to extremely exacting standards and machining tolerances. This results in high component cost. The high cost effect may be mitigated by quantity production of standard systems and components, which is a widespread practise in the diesel industry. Many fuel injection systems are built by specialist firms such as Robert Bosch and Lucas CAV, which offer a range of pumps and nozzles applicable to many types and sizes of engine.

The air blast system consists of a two-part metering orifice fitted in the engine cylinder. One orifice controls the fuel and the other the air, the air blast being admitted around the fuel jet so as to entrain it in droplet form and propel it in to the combustion space. An air compressor is obviously required, and in the automobile application this consists of a simple piston-type unit driven from a convenient point. Most heavy vehicles are already fitted with a suitable compressor unit for air brake operation, and a simple increase in compressor capacity would provide for the fuel injectors. The third major element is the microprocessor control system and its associated sensors and input and output actuators. This is comparable in sophistication and configuration to the units used in current automobiles. None of these components requires unusual levels of design or manufacturing capability. A three cylinder gasoline automotive engine developing 100bhp, complete with three injectors, an air compressor and control system, can be built for approximately the cost of the injection system alone for a diesel engine of the same power.

A diesel fuel injection system is required to provide a very high degree of control over the process of spraying the fuel into the combustion chamber. This is crucial to engine performance, fuel economy and exhaust emissions. The air blast system at its current level of development is not capable of providing the same control over fuel distribution and penetration, and may prove to be inherently incapable of such precise action. However, it is possible that methanol fuelling may not require it, because methanol vaporizes so much more readily than diesel. The balance of fuel charge control is therefore tipped towards air movement and swirl patterns, and the importance of the ability of an injector to spray the incoming fuel to a given location is greatly diminished.

One method of diagnosing diesel injector condition is to observe the engine exhaust fumes. Excessive black smoke will often indicate an injector which is releasing fuel at too low a pressure or failing to seal after injection. This type of diagnosis may be regarded as crude in the light of modern emissions concerns, and there are other engine faults which may produce the same symptoms. Evidence from MILE and other methanol field trials has shown that methanol injectors can deteriorate to levels which would produce drastic smoking in a diesel without any corresponding external evidence of their condition. So far as is known, no attempt has yet been made to quantify the relationship between methanol injector condition and non-visible exhaust emissions. Some research work on injector life undertaken during the early part of project MILE showed that methanol injectors operated on a special test rig failed routine pressure and flow rate checks after test durations far shorter than actual in vehicle service times. Few injectors lasted more than 50 hours on the test rig, which simulated engine operation at a steady 1,000 rpm. This led to the speculation that many of the injectors in working vehicles were in fact unserviceable by normal diesel criteria, but still capable of providing acceptable performance and fuel economy together with no visible exhaust emissions. This reasoning led in turn to the speculation that perhaps the requirements of methanol injection were less exacting, and could be met by a simpler type of injector unit.

V CONCLUSIONS

The low-pressure air blast injector can be used to inject methanol fuel into a conventional heavy-duty two-stroke engine.

It would be advantageous to use a higher air blast pressure than could be provided by the semi-production injector used. The principle of operation and construction could easily be adapted to significantly higher blast pressures, which would offer wider latitude in scheduling the fuel injection timing and flow rate.

The introduction of methanol fuel into the cylinder prior to the requisite ignition point caused far more problems than anticipated. It appears that some considerable modification to the engine would be necessary to eliminate pre-ignition throughout the operating range.

The data obtained during the project testing are insufficient to draw any conclusions regarding engine performance and emissions levels, or injector reliability. It became apparent that the air-blast injection technology as applied in a two-stroke engine is a complex and multi-variable phenomenon, which cannot be applied casually to an existing engine design. The interactions between air flows, scavenging, charge stratification, injection timing, injection duration and spark timing are not amenable to simple first order estimates as to variable settings. Only through long term continued exposure and empirical testing would some relationships begin to present themselves, and then only in a carefully controlled environment designed to investigate these relationships.

The air blast injector is considerably easier and cheaper to manufacture than a conventional diesel unit injector or pump-line-nozzle combination in a production situation.

Many of the problems which prevented effective engine testing during the project were individually not important, and could easily be overcome by routine design and engineering measures in a production situation.

VI RECOMMENDATIONS

1. Reconfirm and expand the relationship established with the injector manufacturer during the course of the project.

2. Continue with work to perfect operation of engine/injector combination, including cooling of incoming air charge, modification of piston and combustion chamber shape, improvements in engine cooling and other measures necessary to eliminate pre-ignition.

3. Carry out theoretical feasibility study on installation of the injector in a four-stroke engine.

4. Examine the feasibility of raising the operating air pressure of the injector in order to provide greater freedom of injection event timing.

5. Examine the feasibility of using low pressure mechanical injectors in a similar type of engine.

Exhibit 1. Injector Test Matrix

TERA 2-71 TEST MATRIX

June 8, 1990

Variables:	Speed	Torque	Ign. Timing	Injection Timing				Blower Pos.
				Fill Start	Fill Duration	Inj. Start	Inj. Duration	
	rpm	nm	c.a.	c.a.	c.a. *	c.a.	c.a. *	% open
Range:	600 to 2000	0 to 249	300 atdc to 359 atdc	270 atdc to 125 atdc	0 to 215 to achieve Torque	110 atdc to 285 atdc	0 to 175 to achieve Torque	100 to 0

SENSITIVITY TEST MATRIX

#		1	2	3	4	5	6	7
Preset Speed & Torque Pts.	rpm	600	1300	1300	1300	2000	2000	2000
	nm	0	5	125	249	4	100	200

At each speed/torque point do:	Ign. Timing	Inj. Start	Blower Pos.
	MBT	Max. Adv.	Max.
	Ign. Limit	Max. Ret.	Min.
	Mid Adv	Center	Mid.
	Part. Burn	Mid Adv	(Very only
	Mid Ret	Mid Ret (@ MBT)	@ MBT and Mid. Ret.)

Total Test Points for Sensitivity Test is 77 points.
From sensitivity test select optimum points based on efficency.
Then run a 13 mode test with emissions at the above determined test.

Ign. Timing
- MBT is the minimum spark advance for best torque.
- Ignition Limit is the maximum spark advance which just starts misfiring.
- Partial Burn is the maximum spark retard which just starts misfire.

Inj. Timing
- Max. Adv. is the maximum injection advance which gives 10% peak torque drop.
- Max. Ret. is the maximum injection retard which gives 10% peak torque drop.
- Center is the maximum torque injection timing between Max. Adv. and Max. Ret.

Blower Pos.
- Mid. is the maximum torque blower position.
- Max. is the maximum blower position at 10% torque drop.
- Min. is the minimum blower position at 10% torque drop.

Exhibit 2. Injector Test Data Sheet

Test #	1	2	3	4	5	6	7	8	9
Speed (rpm)	850	1130	1138	1347	1425	1430	1456	1806	1802
Torque (ftlbf)	4	32	120	122	112	62	82	102	46
Fuel Start (gm)		2900	3000	3580	3600	3800	3800	4000	3800
Fuel End (gm)		2400	2500	2600	3000	3300	2600	2800	2800
Fuel Time (s)		156	120	201	130	175	226	173	173
Air Delta P (in H2O)		0.10	0.14	0.19	0.08	0.06	0.11	0.19	0.18
Air In P (in H2O)		0.13	0.13	0.20	0.20	0.40	0.13	0.20	0.18
Air Box P (psig)		0.90	0.90	1.05	1.00	0.55	0.50	1.10	1.10
Exhaust P (in H2O)		0.80	0.80	1.00	0.30	0.45	0.35	1.80	1.30
Atm. P (in Hg)		29.6	29.6	29.5	29.6	29.6	29.5	29.5	29.5
Air Box T (f)		175	180	202	170	175	183	254	210
Coolant T (f)	190	184	184	177	182	184	173	177	175
Fuel T (f)				103			102	101	
Oil T (f)				216			210	221	
Cyl 1/2 T (f)		119/760	719/775	824/822			863/932	1000/100	850/290
Air In T (f)									
Amb. Dry T (f)	78	80	82	84	84	84	81	83	83
Amb. Wet T (f)									
Inj. P.W. (msec)	1.5	5.0	5.0	5.0	5.0	4.0	4.0	4.0	
Inj Angle (atdc)	150	150	150	177	177	177	178	230	
Fill P.W. (msec)	6.0	12.6	12.6	16.0	12.0	9.5	12.0	16.0	
Fill Angle (atdc)	350	330	330	330	330	330	330	330	
Ign Angle (atdc)	-40	-35	-30	-40	-25	-15	-20	-30	
Fuel/Air P (psi)	90/80	89/72	89/70	90/80	85/60	88/70	90/80	90/80	
Norland disk #				114	100	104	113	115	

Power (hp)	1	7	26	31	30	17	23	35	16
Efficiency (%)		8	23	24	25	22	16	19	10
Airflow (cfm)	13	51	63	76	45	35	53	77	74
Airflow (lbm/hr)	60	233	290	351	206	159	245	352	342
Fuelflow (lbm/hr)		25.4	33.1	38.7	36.6	22.7	42.1	55.1	45.9
Scavenging Ratio		1.3	1.6	1.7	0.9	0.7	1.1	1.3	1.2
Charging Eff		0.76	0.82	0.83	0.64	0.55	0.71	0.78	0.75
Trapping Eff		0.60	0.52	0.50	0.72	0.78	0.65	0.58	0.61
Trapped A/F Ratio		5.5	4.5	4.5	4.1	5.5	3.8	3.7	4.6
Lambda		0.8	0.7	0.7	0.6	0.8	0.6	0.6	0.7

Fuel Energy Content J/gm ------	20000	Fuel Stioch A/F Rat	6.47

Air Density @ 59 deg F, 14.7 psi lbm/ft^3	0.0765	Ideal Cyl. Disp @ Stand. atm. ft^3	0.0410
Standard Temp F	59		
Standard Press. in Hg	30		

Exhibit 3: Cylinder Head Cross Section of 2 71 Engine with Injector, Spark Plug and Combustion Chamber Pressure Transducer

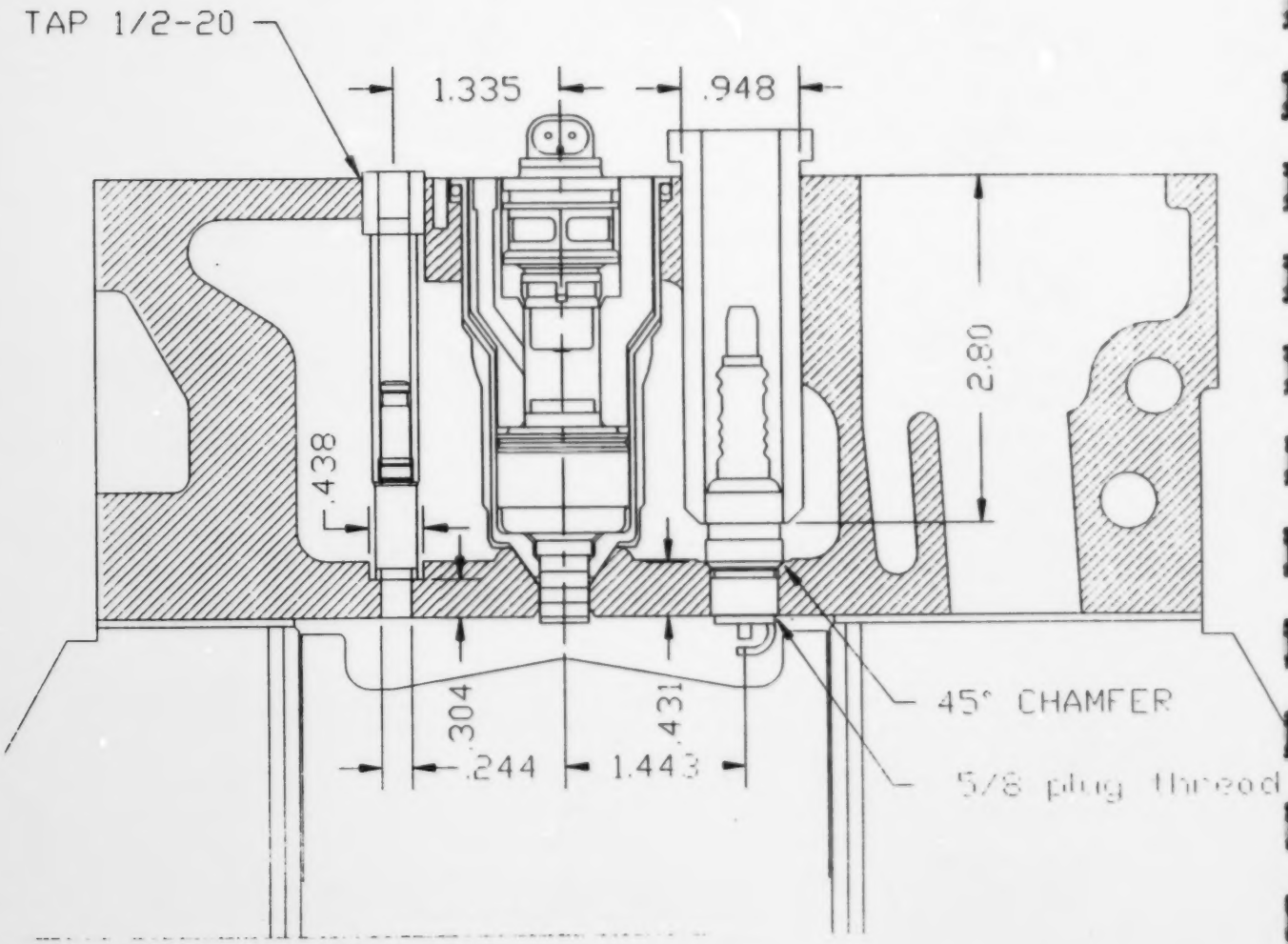


Exhibit 4. Injector Water Jacket

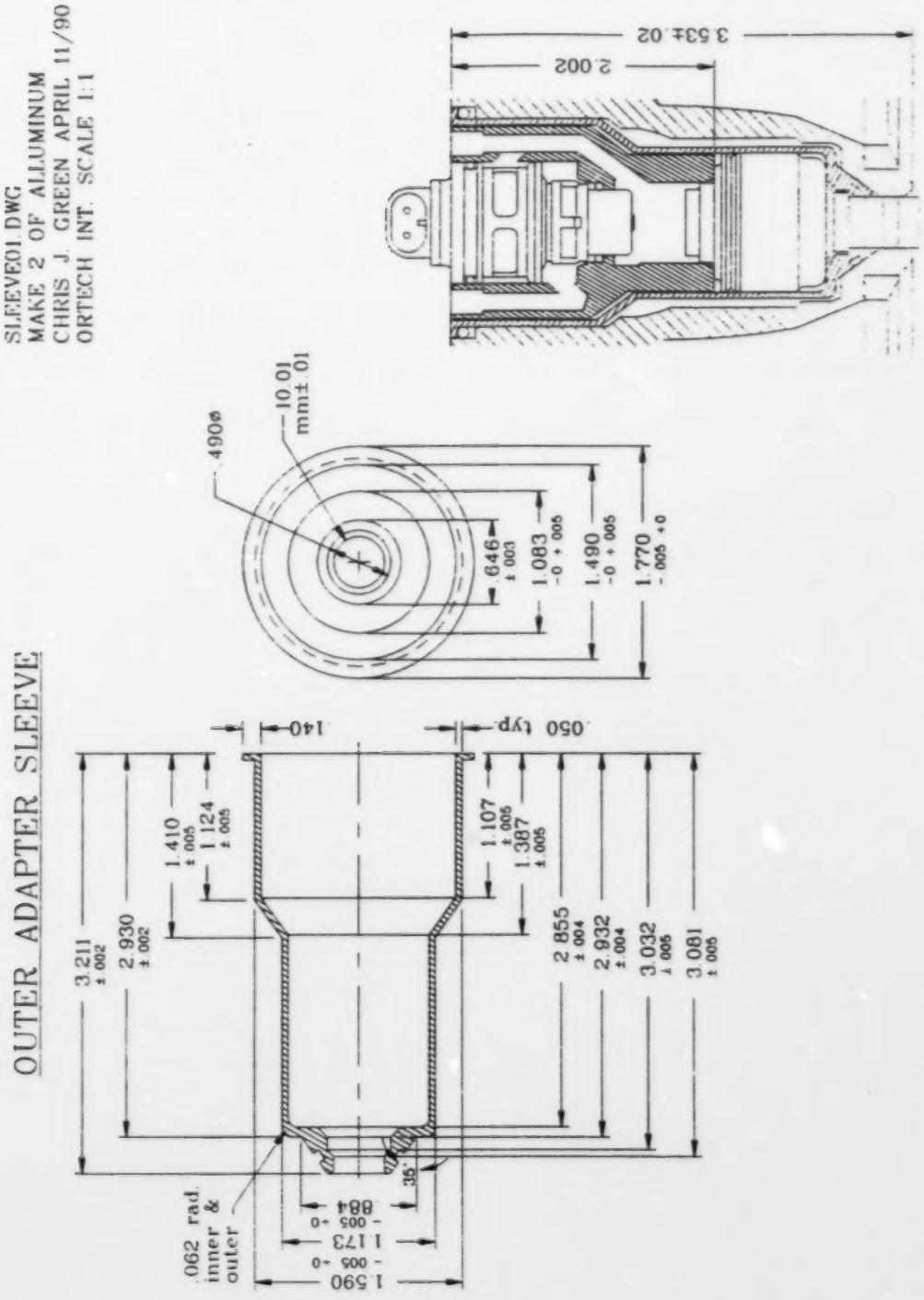


Exhibit 5. Injector Sleeve

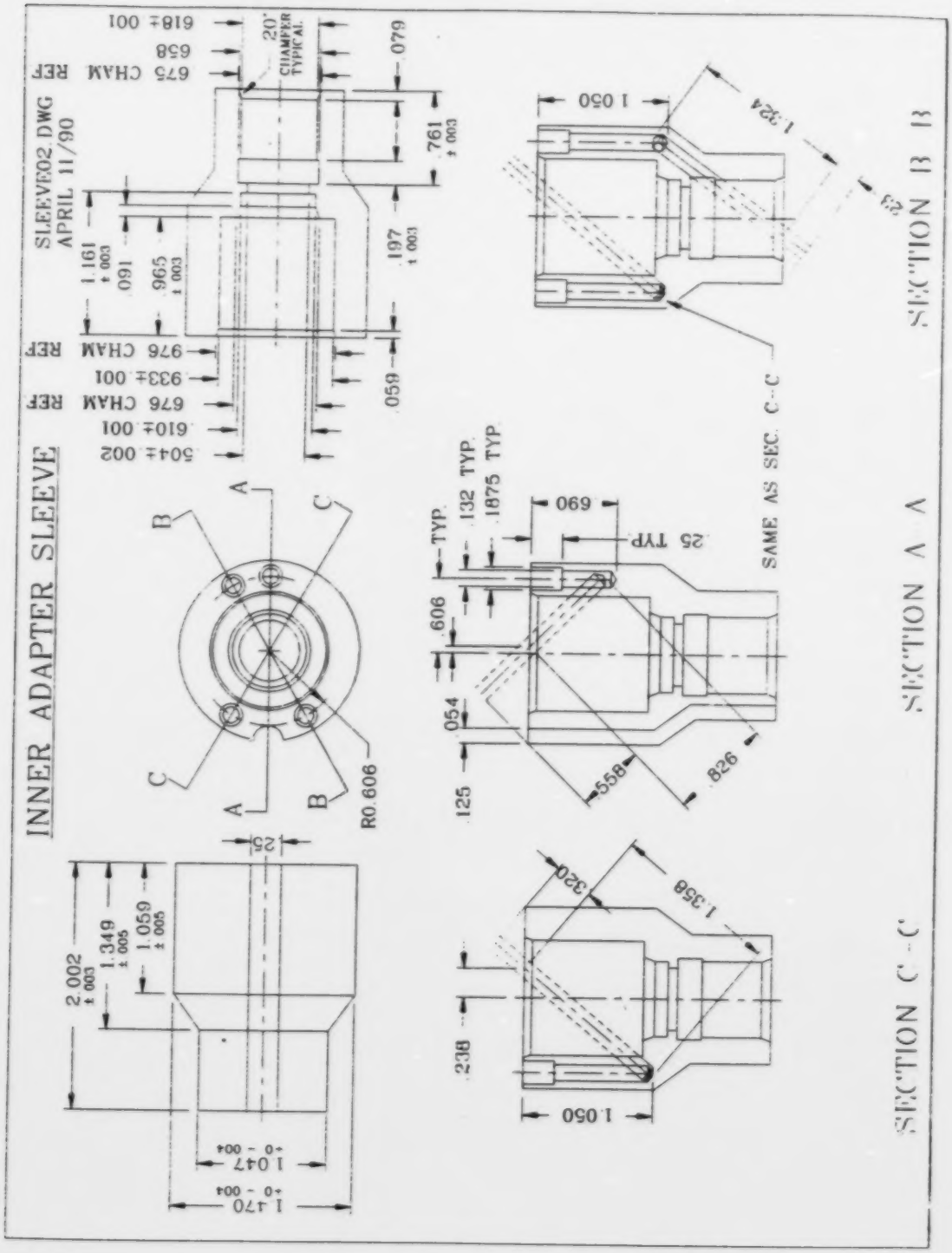


Exhibit 6. Top View of Cylinder Head with Injector and Hold-Down Clamps

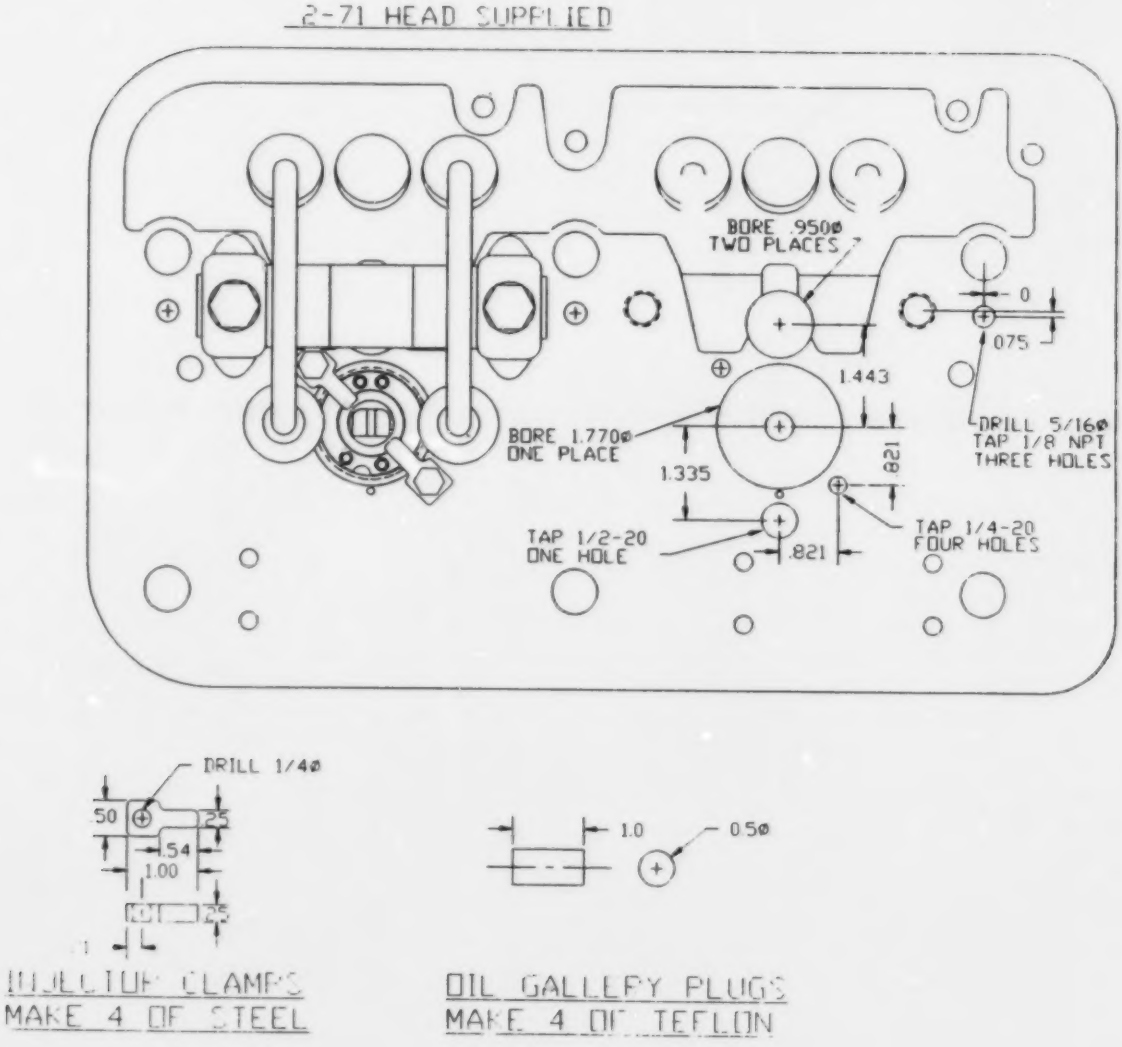


Exhibit 7. Top View of Cylinder Head with Supply Lines of Fuel, Air and Wiring Installed

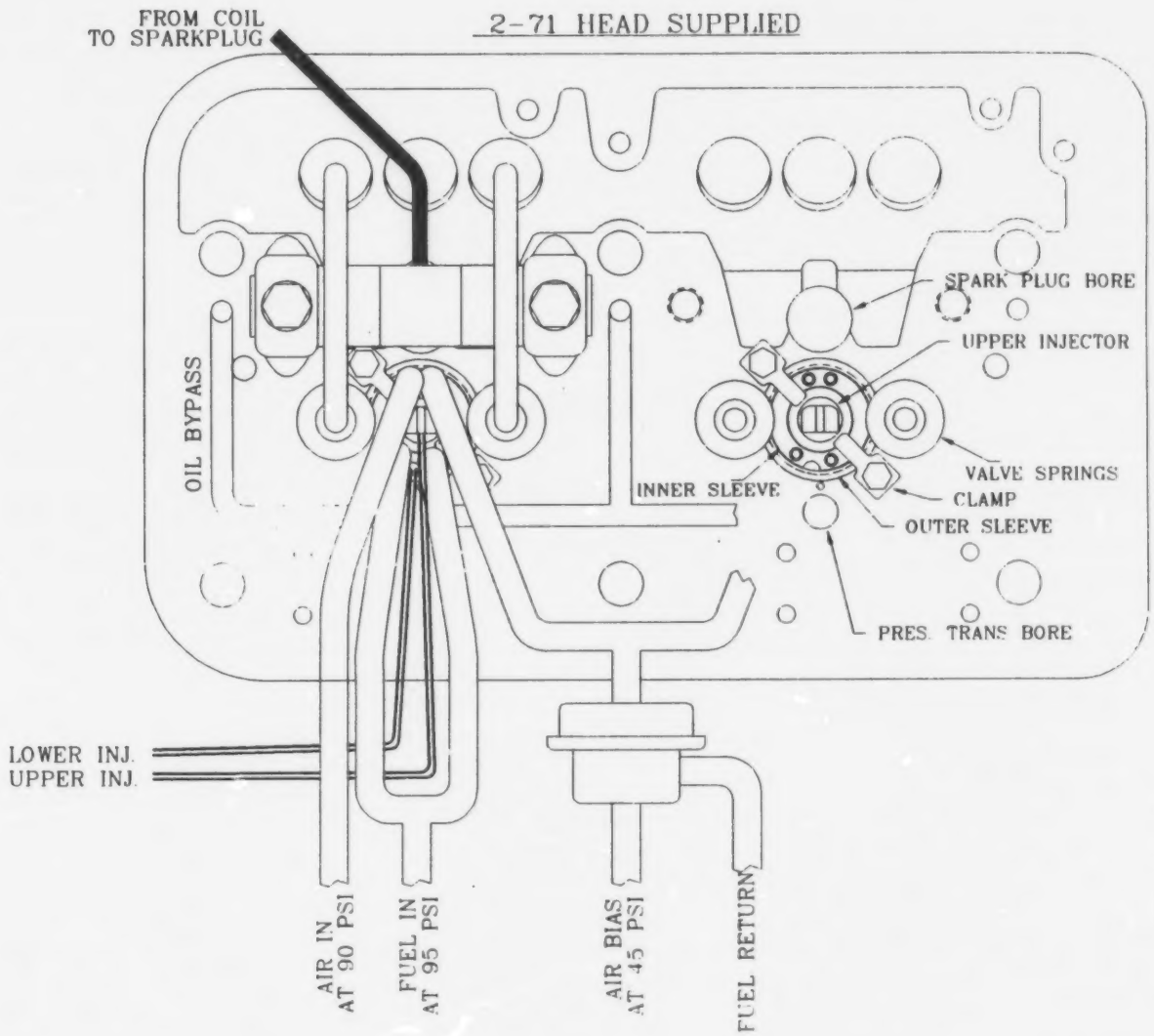


Exhibit 8. Test Cell Electric Schematic for DDC 2-71 with Air Assisted Injector

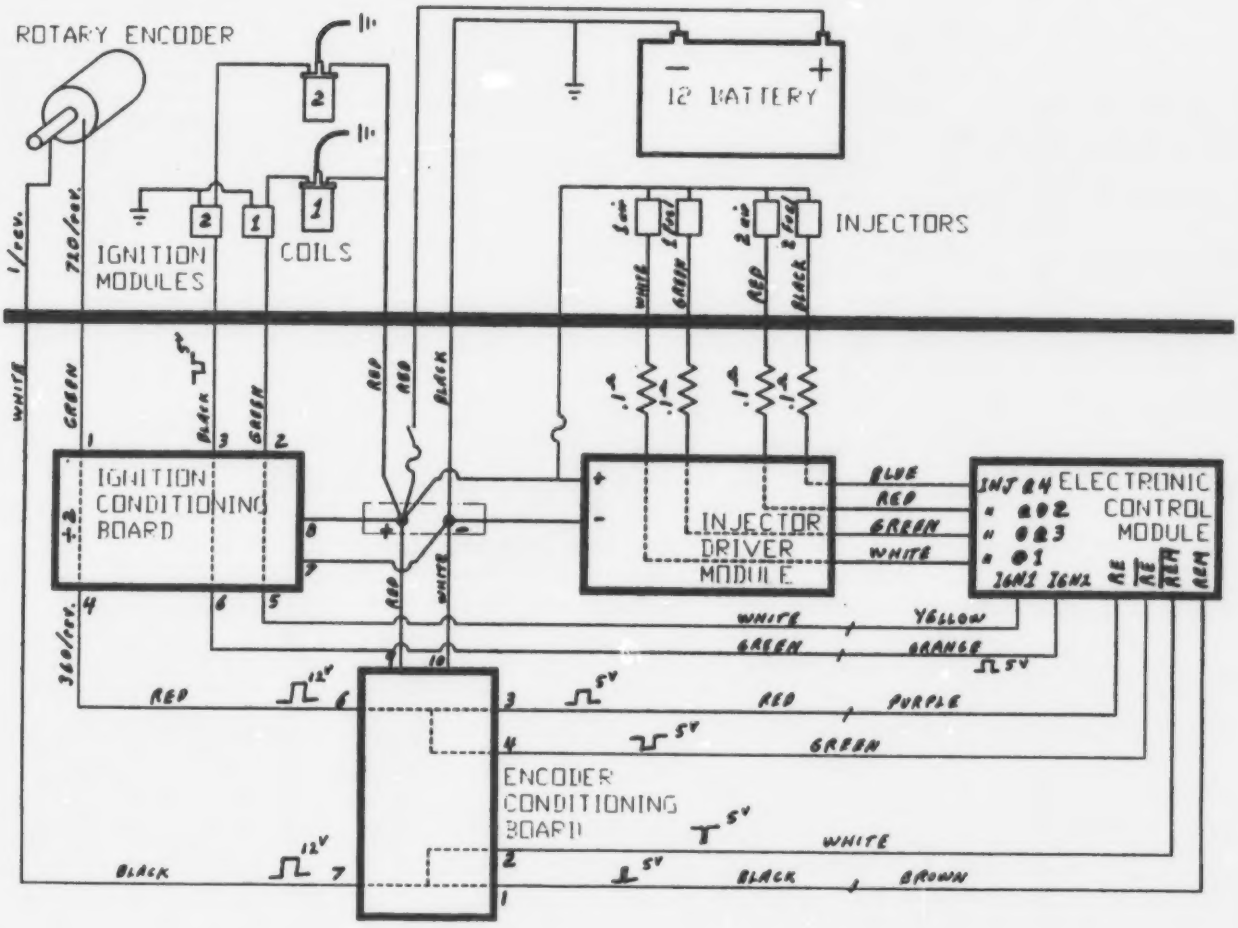


Exhibit 9. Cylinder Pressure Traces for Test Points #3 and #7

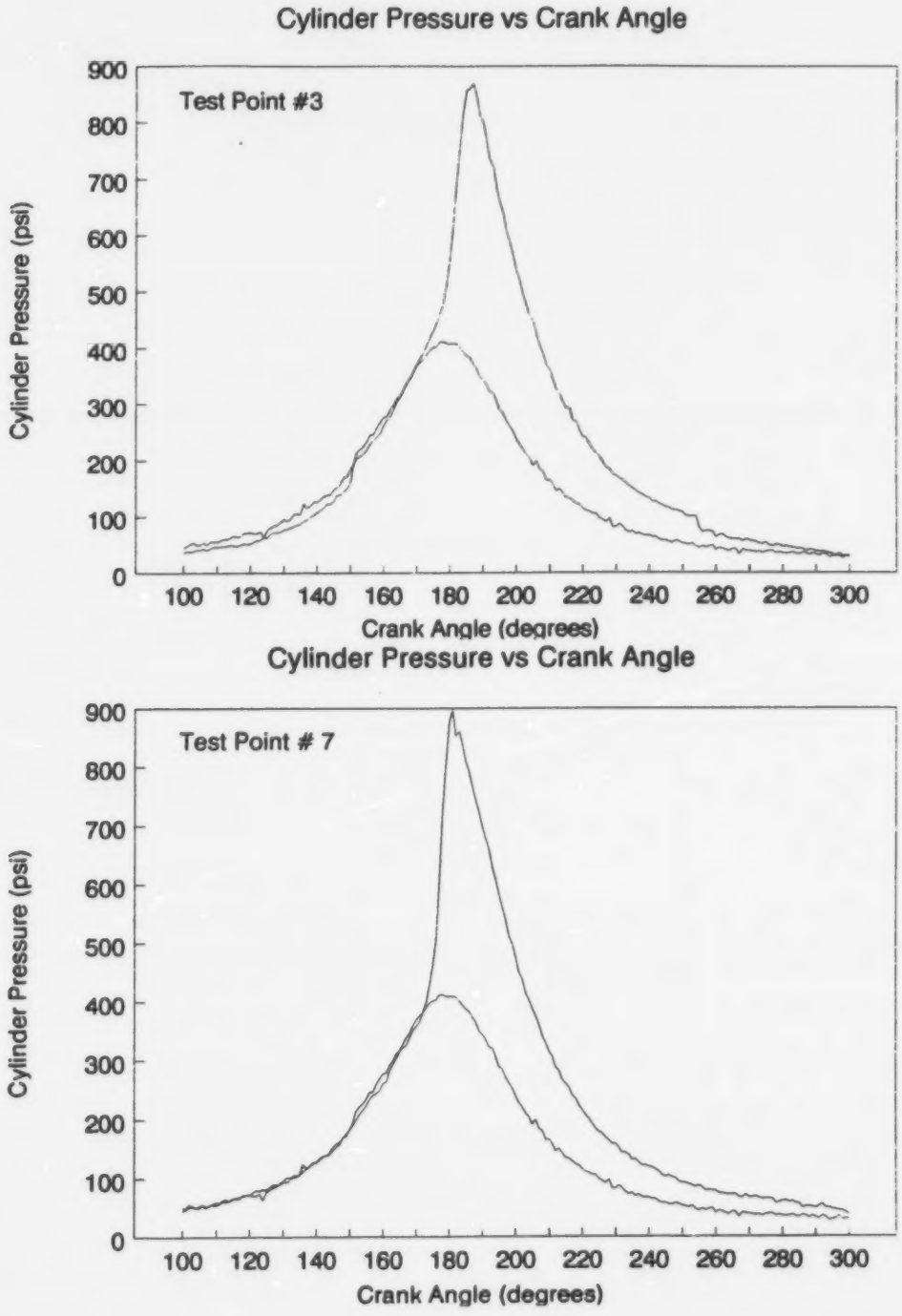
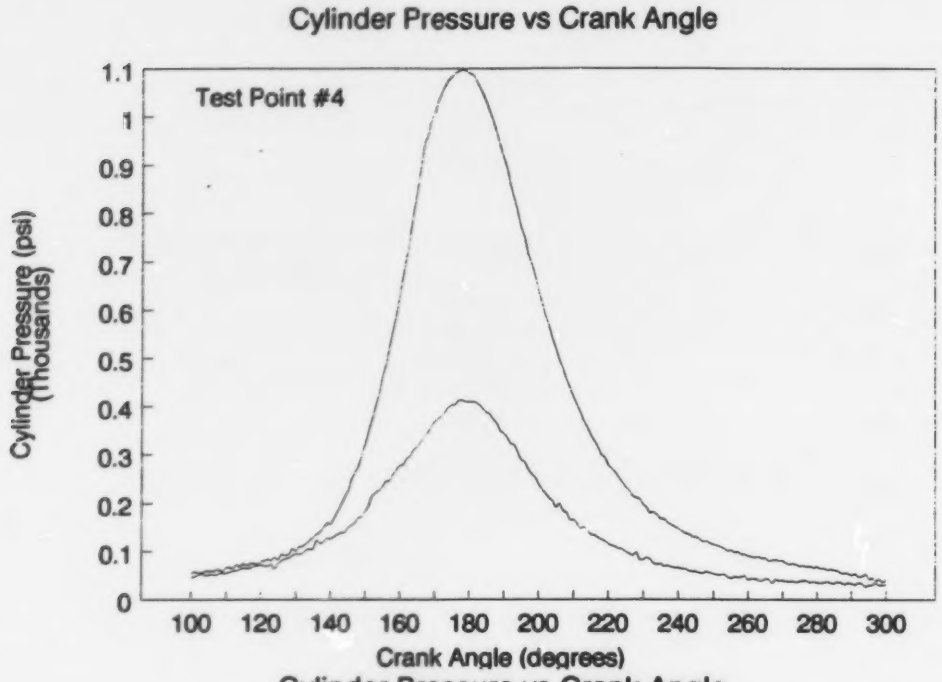


Exhibit 10. Cylinder Pressure Traces for Test Points #4 and #8



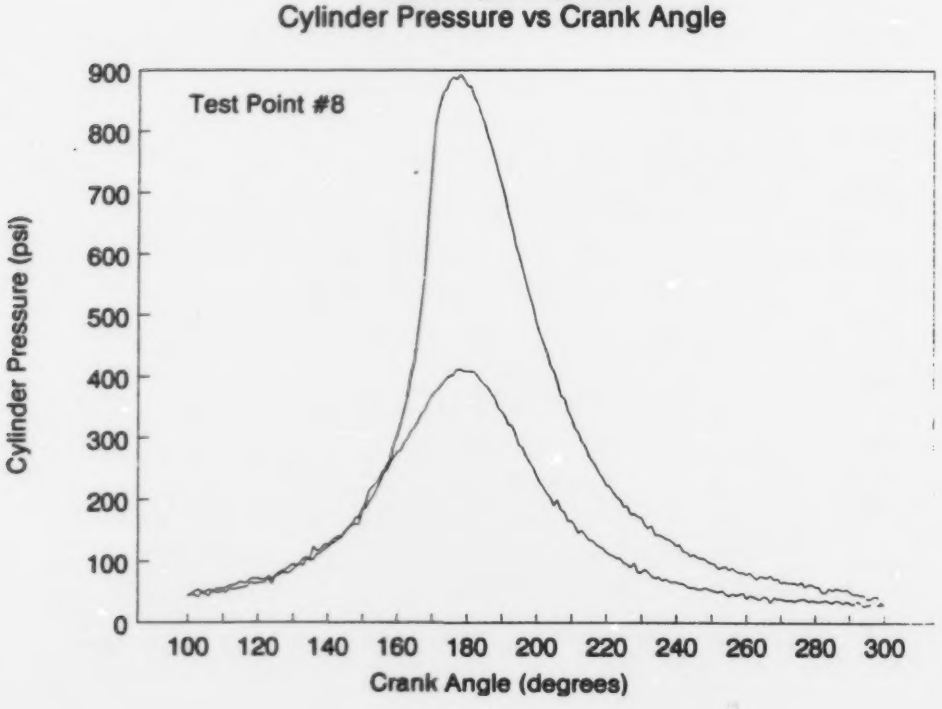
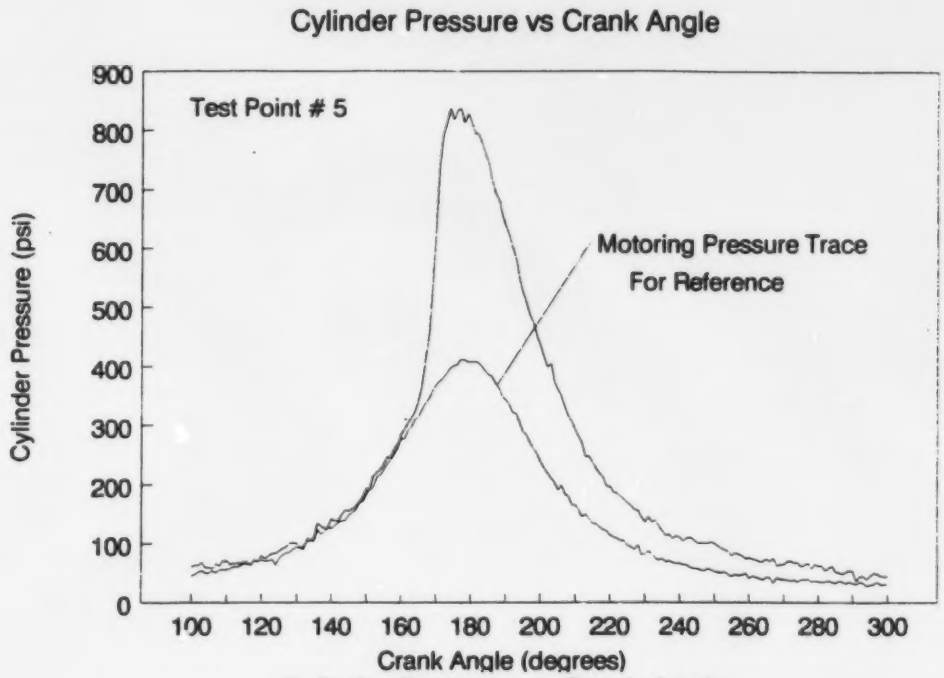
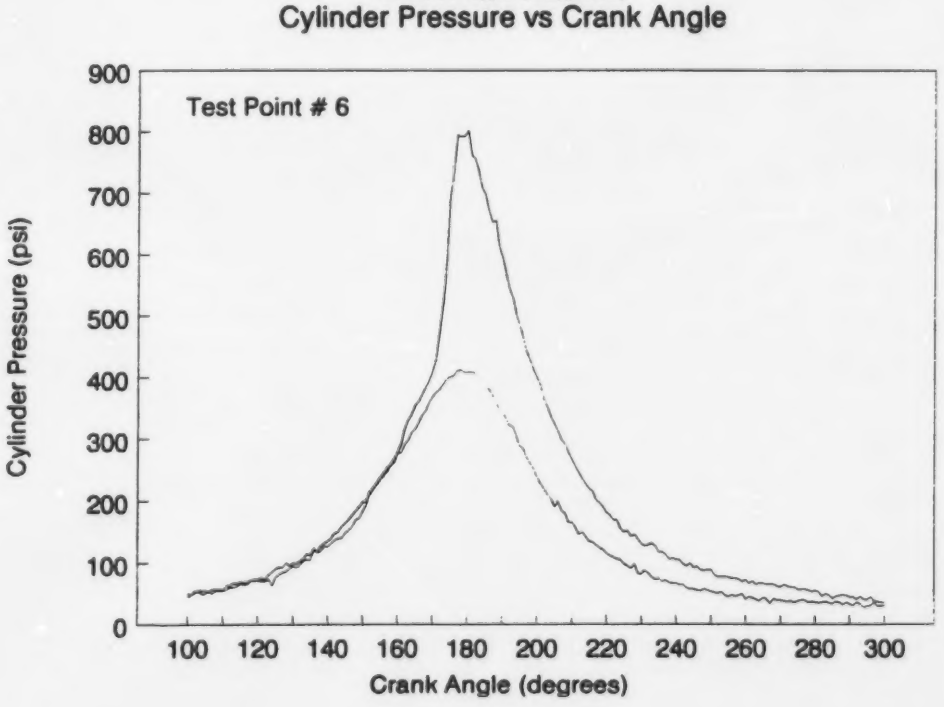


Exhibit 11. Cylinder Pressure Traces for Test Points #5 and #6





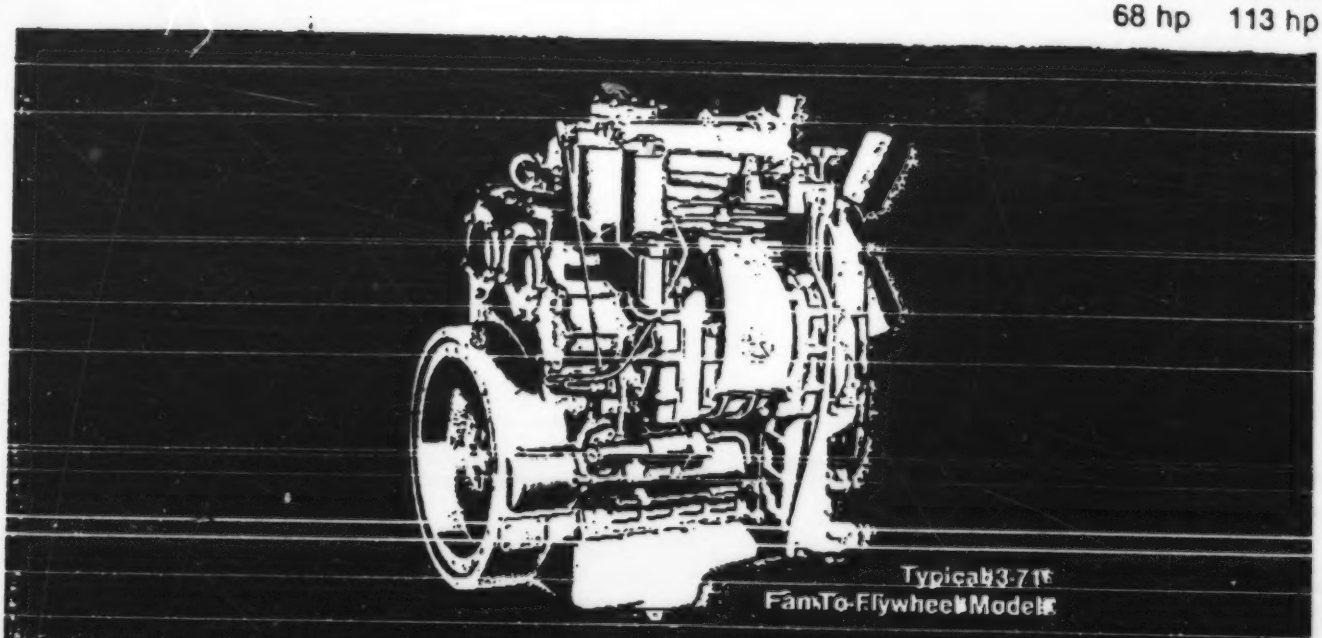
APPENDIX A

SPECIFICATIONS AND RATINGS OF 2-71 DDC ENGINE

Detroit Diesel Engines

fan-to-flywheel models

2-71 68 hp 3-71 113 hp



specifications

Basic Engine	2-71 N70 injectors	3-71 N65 injectors
Model	1023-5100	1033-5000, 1033-7000
Description	Naturally Aspirated	Naturally Aspirated
Number of Cylinders	2	3
Bore and Stroke	4.25 in X 5 in (108 mm X 127 mm)	4.25 in X 5 in (108 mm X 127 mm)
Displacement	142 cu in (2.33 liters)	213 cu in (3.49 liters)
Rated Gross Power: 60°F (15.6°C) and 29.92 In Hg (101.31 kPa) Barometer (Dry)	68 BHP (51 kW) @2000 RPM	113 BHP (84 kW) @2100 RPM
85°F (29.4°C) and 29.00 in Hg (98.19 kPa) Barometer (Dry)	65 BHP (48 kW) @2000 RPM	109 BHP (81 kW) @2100 RPM
Continuous Gross Power: 85°F (29.4°C) and 29.00 In Hg (98.19 kPa) Barometer (Dry)	48 BHP (36 kW) @1500 RPM	82 BHP (61 kW) @1800 RPM
Torque: 85°F (29.4°C) and 29.00 In Hg (98.19 kPa) Barometer (Dry)	184 lb ft (249 N•m) @1400 RPM	299 lb ft (405 N•m) @1400 RPM
Compression Ratio	17 to 1	19.7 to 1
Approximate Dimensions: Length	32 in (813 mm)	36 in (914 mm)
Width	27 in (686 mm)	29 in (737 mm)
Height	41 in (1041 mm)	41 in (1041 mm)
Net Weight (Dry)	960 lbs (435 kg)	1525 lbs (692 kg)

For complete dimensional information, refer to the detail drawing 25A23 for Model 1023-5100, 25A75 for Model 1033-5000, and 25A73 for 1033-7000.

performance

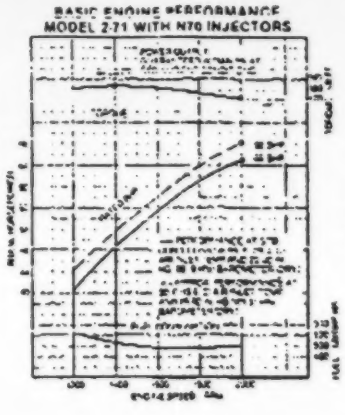
Rating Explanation

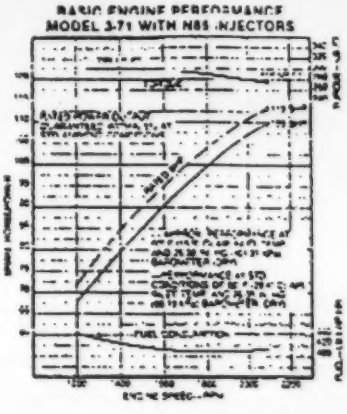
RATED BHP is the power rating for variable speed and load applications where full power is required intermittently.

CONTINUOUS BHP is the power rating for applications operating under a constant load and speed for long periods of time.

FUEL CONSUMPTION CURVE shows fuel used in pounds per brake horsepower hour.

THESE RATINGS do not include power requirements for accessory and standard equipment.





standard equipment

Air Inlet Housing

Alternator—12 volt, 42 amp

Crankshaft Pulley

Engine Mounts

Exhaust Manifold

Fan—18 in (457 mm), 6-blades, suction

Flywheel—SAE #2, 2-71 only; SAE #1, 3-71 only

Flywheel Housing—SAE #2, 2-71 only; SAE #1, 3-71 only

Fuel Filters

Governor—Constant speed, 2-71 only; variable speed, 3-71 only

Injectors—Cam operated, unit type, clean tip

Instruments—Ammeter, oil pressure and water temperature gauges, and starter switch

Lube Oil Cooler

Lube Oil Filter

Oil Pan—Stamped steel pan with 16° inclination angle

Starting Motor—12 volt

For complete specifications and a listing of standard and optional equipment, consult your authorized Detroit Diesel Allison representative.

Specifications subject to change without notice.

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